



A STUDY OF SUPER-ADIABATIC COMBUSTION ENGINE

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Abstract—A one-dimensional numerical calculation has been performed on a reciprocating heat engine with super-adiabatic combustion in porous media. The system consists of two pistons and a thin porous medium in a cylinder; one being a displacer piston and the other a power piston. These create reciprocating motions with a phase relation, similar to those of an ordinary Stirling engine. By means of the reciprocating flow system, the residual combustion gas enthalpy is effectively regenerated to induce enthalpy increase in the mixture through the porous medium, which provides heat storage. The results show that due to heat recirculation, the maximum temperature is higher than the theoretical one in free space. Furthermore, using a porous medium with a high absorption coefficient, since the thermal diffusion by radiation decreases in the porous medium, the maximum temperature increases, resulting in the extension of the flammability limit for the heat engine. © 1997 Elsevier Science Ltd.

Super-adiabatic combustion Porous medium Heat engine Reciprocating flow

NOMENCLATURE

- A = Frequency factor for combustion (s^{-1})
- A_p = Surface area of an equivalent particle forming porous medium (m^2)
- C_v = Specific heat at constant volume ($kJ\ kg^{-1}\ K^{-1}$)
- C_s = Specific heat of solid phase ($kJ\ kg^{-1}\ K^{-1}$)
- d = Diameter of cylinder (m)
- d_p = Equivalent diameter of a particle forming porous medium (m)
- D = Diffusion coefficient ($m^2\ s^{-1}$)
- Da = Darcy number (equation (13))
- E = Activation energy ($kJ\ mol^{-1}$)
- h_o = Heat of combustion ($kJ\ kg^{-1}$)
- H_o = Dimensionless heat of combustion (equation (13))
- H_{tot} = Dimensionless over-all heat transmission (equation (13))
- HR = Dimensionless heat release rate (equation (13))
- k_{eq} = Permeability of porous medium (m^2)
- K = Overall heat transmission coefficient for heat loss ($Wm^{-2}\ K^{-1}$)
- L_s = Dimensionless stroke (equation (13))
- M = Dimensionless heat transfer coefficient between two phases (equation (13))
- n = Number of moles of working gas
- n_p = Number density of equivalent particle forming porous medium (m^{-3})
- Nr = Radiation-conduction parameter (equation (13))
- p = Pressure (Pa)
- r = Crank radius (m)
- r_c = Thermal conductivity ratio (equation (13))
- Re = Reynolds number (equation (13))
- R = Gas constant ($kJ\ mol^{-1}\ K^{-1}$)
- t = Time (s)
- T = Temperature (K)
- u = Velocity (ms^{-1})
- W = Reaction rate ($kg\ m^{-3}\ s^{-1}$)
- x = Coordinate system (m)
- x_o = Thickness of porous medium (m)
- X = Dimensionless distance (equation (13))
- Y = Product mole fraction

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Greek symbols

- α_s = Heat transfer coefficient between gas and porous medium ($\text{Wm}^{-2} \text{K}^{-1}$)
 Γ = dimensionless heat capacity ratio between two phases
 ϵ_1 = Emissivity of porous medium
 ϵ_2 = Emissivity of wall
 θ = Dimensionless temperature
 θ_c = Crank angle (degree)
 κ = Absorption coefficient (m^{-1})
 λ = Thermal conductivity of gas phase ($\text{Wm}^{-1} \text{K}^{-1}$)
 λ_s = Thermal conductivity of porous medium ($\text{Wm}^{-1} \text{K}^{-1}$)
 μ = Viscosity (Pa s^{-1})
 ρ = Density of gas phase (kg m^{-3})
 ρ_s = Density of porous medium (kg m^{-3})
 σ = Stefan-Boltzmann constant ($\text{Wm}^{-2} \text{K}^{-4}$)
 ψ = Phase relation angle between two pistons (degree)
 ω = Angular velocity (s^{-1})

INTRODUCTION

A chemical energy, such as a fossil fuel, is usually converted into mechanical and/or electric powers through combustion. In this case, though the combustion efficiency is close to 100%, the thermal efficiency and the effectiveness are not so high because of the irreversible process in combustion. On the other hand, it is well known that both the thermal efficiency and the effectiveness increase with temperature of the system. Consequently, excess enthalpy burning is a useful method for decreasing irreversibility of combustion and creating a high temperature flame, which will be successfully applied to a new energy conversion method.

Recently, super-adiabatic combustion with reciprocating flow in a porous medium was investigated in an experiment [1] and a numerical calculation [2]. In this system, a combustible gas with an extremely low heat content flows into a porous medium where the flow direction reverses regularly. Using the reciprocating flow system, the combustion gas enthalpy is effectively regenerated to induce an enthalpy increase in the combustible gas through the porous medium, which provides heat storage. The results revealed that the flame temperature in the porous medium is 13 times higher than the theoretical flame temperature of an ordinary flame in free space. Moreover, the heating value of the combustible gas is equivalent to a temperature increase of only 50 K. On the basis of this novel combustion technology, a new heat engine has been proposed [3]. Through a numerical calculation, it is revealed that the thermal efficiency of the heat engine is higher than the conventional Otto and diesel cycles under the condition of the low compression ratio.

The present study shows the combustion process and thermal efficiency of the reciprocating heat engine with the super-adiabatic combustion and shows the effects of the heat loss and radiative transfer to the thermal efficiency and flammability limit of the heat engine.

CYCLE OF SUPER-ADIABATIC COMBUSTION ENGINE

Figure 1 shows a schematic diagram of the cycle of a reciprocating heat engine capable of super-adiabatic combustion in a porous medium. It consists of two pistons and a thin porous medium in a cylinder. One is a displacer piston and the other a power piston. These create reciprocating motions with a phase relation, similar to those of an ordinary Stirling engine. In the space between the displacer piston and the porous medium, first, a fresh mixture displaces the combustion gas in the cylinder (i.e. scavenging stroke (Fig. 1 ①)). Then, it is compressed, similarly to conventional two-stroke internal combustion engines when the power piston is located in the vicinity of the porous medium (Fig. 1 ②). After the compression stroke, both pistons move in the same direction with almost the same velocity. As a result, the compressed mixture is preheated and burns in the porous medium; that is, constant-volume combustion is achieved (Fig. 1 ③). The solid line in the porous medium presents a schematic profile of the temperature for both the gas and solid phases. Combustion heat is converted into mechanical power in the expansion stroke when the displacer piston remains in the vicinity of the porous medium (Fig. ④). On reversal, the residual enthalpy of combustion gas is effectively accumulated by the porous medium (Fig. 1 ⑤).

By means of the reciprocating motions, the residual combustion gas enthalpy is effectively regenerated to induce enthalpy increase in the compressed mixture through the porous medium, which provides heat storage. The cycle proposed here is similar to that of an internal combustion Stirling cycle [4–6], in which liquid fuels are directly injected into the space between the porous medium and the power piston and burn, similarly to conventional diesel cycles. In the present cycle, since constant-volume combustion is achieved, similarly to the Otto cycle, a thermal efficiency higher than that of the previous internal combustion Stirling cycle is expected.

THEORETICAL ANALYSIS

On the basis of the schematic diagram for the heat engine as shown in Fig. 1, we assumed one-dimensional gas flow in a cylinder after a fresh mixture displaces a combustion gas in the cylinder instantaneously. Both pistons are operated by means of a crank-piston system with the ratio of the connecting rod to the crank radius λ_p and with the phase relation angle ψ . The porous medium consists of homogeneously dispersed fine solid particles. The combustion reaction is described by an irreversible first-order isomerization (i.e. reactant $\rightarrow$ product). The porous medium is noncatalytic. The Lewis number is unity. The physical properties are constant except for gas density. In the displacer piston side the surfaces of the cylinder and the piston are maintained at temperature T_0 . On the other hand, in the power piston side an overall heat transfer coefficient is given, including a thermal conductivity of the wall and inner and outer heat transfer coefficients; as a result, the surface temperature of the wall changes in a cycle. In order to clarify the effect of radiation in the porous medium, the diffusion approximation is applied to the analysis for radiative transfer. Furthermore, the radiative heat exchange between the surfaces of the porous medium and walls of the cylinder and piston is taken into account. The volumetric viscous force for the flow is expressed by Darcy's law.

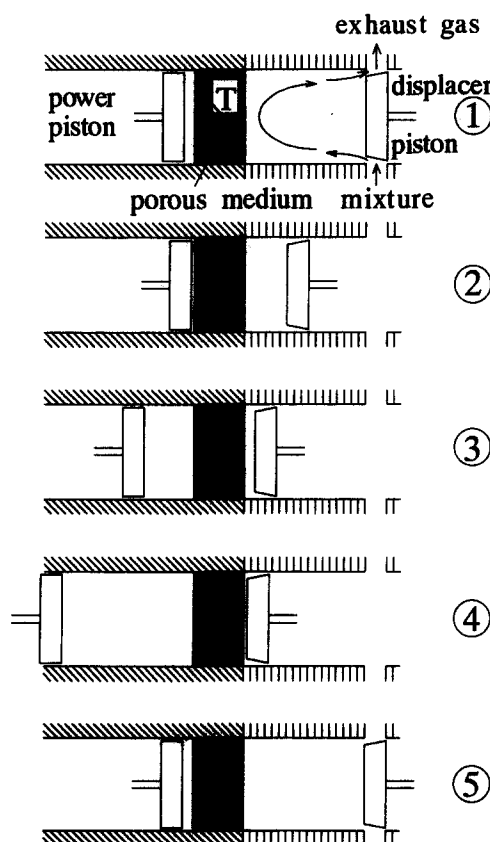


Fig. 1. Schematic diagram of the cycle of a reciprocating heat engine capable of super-adiabatic combustion in a porous medium.

Using the above assumptions, the continuity equation and the conservation equations of momentum, energy and product species in the porous medium are described, as follows.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} = 0 \quad (1)$$

$$\rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} = -\frac{\partial p}{\partial x} + \mu \frac{\partial^2 u}{\partial x^2} - \frac{\mu}{k_{eq}} u \quad (2)$$

$$k_{eq} = \frac{1}{3\pi d_s n_s} \quad (3)$$

$$\rho C_v \frac{\partial T}{\partial t} + \rho C_v u \frac{\partial T}{\partial x} = \lambda \frac{\partial^2 T}{\partial x^2} - p \frac{\partial u}{\partial x} - \alpha_s n_s A_s (T - T_s) + h_o W - \frac{4K}{d} (T - T_o) \quad (4)$$

$$W = A\rho(1 - Y)\exp\left(-\frac{E}{RT}\right) \quad (5)$$

$$\rho_s C_s \frac{\partial T_s}{\partial t} = \lambda_s \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial}{\partial x} \left(\frac{16\sigma T_s^3}{3\kappa} \frac{\partial T_s}{\partial x} \right) + \alpha_s n_s A_s (T - T_s) \quad (6)$$

$$\rho \frac{\partial Y}{\partial t} + \rho u \frac{\partial Y}{\partial x} = D \frac{\partial^2 Y}{\partial x^2} + W \quad (7)$$

$$p = \rho n R T. \quad (8)$$

Herein, ρ and ρ_s are the densities of gas and solid phases, respectively, μ the viscosity of gas, k_{eq} the permeability of the porous medium, d_s the equivalent diameter of a particle, n_s the number density of the particle, λ and λ_s the thermal conductivities of gas and solid phases, respectively, h_o the heat of combustion, α_s the heat transfer coefficient around the particle, A_s the surface area of the particle, A the frequency factor for combustion, E the activation energy, R the gas constant and n the number of moles of the working gas. κ the absorption coefficient, σ the Stefan-Boltzmann constant, d the diameter of the cylinder. The third terms on the right-hand sides in equations (4) and (6) express the heat transfer between two phases. The second term on the right-hand side in equation (6) expresses the radiation transferred similar to the conduction process. The fifth term on the right-hand side in equation (4) expresses the heat loss to the periphery. In this case, in the power piston side the overall heat transfer coefficient is used, while in the displacer piston side the inner heat transfer coefficient is used for K . The heat transfer term in equation (4), the equation (6) and the third term on the right-hand side in equation (2) are eliminated in calculating the velocity and temperature of the gas phase in the cylinder between the pistons and the porous medium. The boundary conditions at the high temperature side and low temperature side surfaces of the porous medium are given, as follows:

$$-\lambda_s \frac{\partial T_s}{\partial x} = \frac{\sigma}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} [T_s^4 - T_w^4], \quad x = \pm \frac{x_o}{2} \quad (9)$$

The location of the piston surface is given by the stroke $2r$ (where r is the crank radius), the ratio of the connecting rod to the crank radius λ_p and the crank angle θ_c , as follows.

$$x = \frac{x_o}{2} + \frac{2r}{2} \left\{ 1 + \cos \theta_c + \frac{1}{\lambda_p} \sin^2(\theta_c + 180) \right\}. \quad (10)$$

The boundary conditions for the gas temperature and product mole fraction are expressed as

follows:

for the displacer piston surface

$$T = T_o, \quad Y = 0; \quad (11)$$

for the power piston surface

$$\partial T / \partial x = 0, \quad \partial Y / \partial x = 0. \quad (12)$$

These equations were transformed into dimensionless forms and were calculated using the variable finite difference method [7]. The principal dimensionless variables and parameters adopted here are as follows:

$$\begin{aligned} X = x/x_o, \quad \theta = T/T_o, \quad U = u/r\omega, \quad L_s = 2r/x_o, \quad Re = r\omega x_o \rho / \mu, \quad H_o = h_o / C_v T_o, \\ Da = k_{eq} / x_o^2, \quad HR = h_o x_o^2 W / (C_v T_o \mu), \quad M = x_o^2 \alpha_s n_s A_s / \lambda, \quad r_c = \lambda_s / \lambda, \\ \Gamma = \rho_s C_s / \rho C_v, \quad Nr = \kappa \lambda / 4\sigma T_o^3, \quad H_{loss} = 4Kd(x_o/d)^2 / \lambda. \end{aligned} \quad (13)$$

For a sponge-like porous material made of cordierite, the porosity is 87.5%. Then, the values of dimensionless parameters are approximately $\Gamma = 300$, $r_c = 1$, $M = 6000$ and $Da = 1 \times 10^{-3}$, when $x_o = 0.01$ m. The specific surface area, i.e. the product of n_s and A_s , of the porous medium is approximately 3000 m²/m³. The heat transfer coefficient α_s is estimated from the heat transfer around a sphere, the diameter of which is equal to the mean diameter of the element forming the sponge-like porous medium. On the other hand, for stainless-steel piled wire meshes, the value of Γ is approximately 3000. Furthermore, the frequency factor and activation energy were estimated assuming that the calculated burning velocity is equal to the measured one at equivalence ratios of 0.53 and 1 for methane–air mixture, resulting in $A = 2.6 \times 10^8$ s⁻¹ and $E = 130$ kJ/mol. Under the condition of the low velocity of the working gas considered here, the pressure drop in the porous medium is negligibly small for $Da \geq 1 \times 10^{-6}$. The overall heat transfer coefficient K was estimated from the inner heat transfer coefficient 400 W/m²K (average in a cycle [8]), the thermal conductivities of Si₃N₄ ceramics (10 mm thickness) and insulator (30 mm thickness) and outer heat transfer coefficient, resulting in 2.34 W/m²K. The emissivities of the porous medium and the wall are 0.2 and 0.8, respectively. The mechanical power is evaluated from the P–V diagram, i.e. the cylinder volume and the arithmetic mean pressure in the cylinder.

RESULTS AND DISCUSSION

Figures 2(a)–(f) show the profiles of the gas and solid temperatures θ , θ_s , the gas velocity U , the reaction rate HR and the product mole fraction Y for the crank angles θ_c of 0, 60, 120, 180, 240 and 300 degree, respectively. Here, the dimensionless stroke L_s is equal to 5, the phase relation angle $\psi = 60$ degree, the dimensionless combustion heat $H_o = 3.2$ and the Reynolds number $Re = 500$, which is equivalent to the speed of 300 rpm. The compression ratio is dependent of the stroke and the phase relation angle. In the present study, the compression ratio is 2. The conduction–radiation parameter Nr ($= \kappa \lambda / 4\sigma T_o^3$) is equal to 10. After scavenging at $\theta_c = 330$, the mixture is compressed in the period over the range of crank angles from $\theta_c = 330$ –120 degree through the B.D.C. (bottom dead center) of the displacer piston. The mixture is preheated and burns in the porous medium, as shown in Fig. 2(c) and (d). The P–V diagram in Fig. 3 indicates that constant-volume combustion is achieved. Furthermore, as shown in Fig. 2(a), the maximum temperature in the porous medium is higher than the theoretical flame temperature θ_{ad} evaluated on the basis of the inlet temperature. Since the combustion continues up to the crank angle of 210 degree, the working gas is almost isothermally expanded over the range from $\theta = 180$ –210 degree. As a result, the combustion heat is converted into mechanical power, first, during the isothermal expansion and then during the isentropic expansion, as shown in Figs 2(d) and (e) and 3.

With increasing conduction–radiation parameter Nr , the maximum temperature in the porous medium increases and the temperature profiles of gas and solid phases become steep because of the decrease in the effect of thermal diffusion by radiative transfer. As a result, the reaction zone is stabilized at the location further to the displacer piston side surface of the porous medium.

As shown in Fig. 3, the profiles of P-V diagram for the cases with and without heat loss are quite similar, though the heating values of those are not the same. The thermal efficiencies for those are 32% ($H_o = 2.7$, $Nr \rightarrow \infty$, without heat loss) and 26% ($H_o = 3.2$, $Nr = 10$, with heat loss), respectively. In the latter case, the ratio of the heat loss to the combustion heat is 38%, the most of which is through the cylinder wall in the displacer piston side. As a result, there is not so large difference between those thermal efficiencies because of a small heat loss in the power piston side; that is, the flowing enthalpy after expansion is divided just into the heat loss and exhausting gas enthalpy in the displacer piston side.

As shown in Fig. 2, the wall temperature in the power piston side T_{w2} varies in a cycle. Figure 4 shows the variation of the wall temperature in a cycle. The wall temperature has the maximum at the crank angle of about 180 degree, in which the reaction rate also reaches the maximum, as shown in Fig. 2. The maximum wall temperature in a cycle reaches the value of 5.3 higher than the porous medium temperature because of the compression of the working gas in the power piston side cylinder. However, the radiant energy emitted from the wall is absorbed by the porous medium

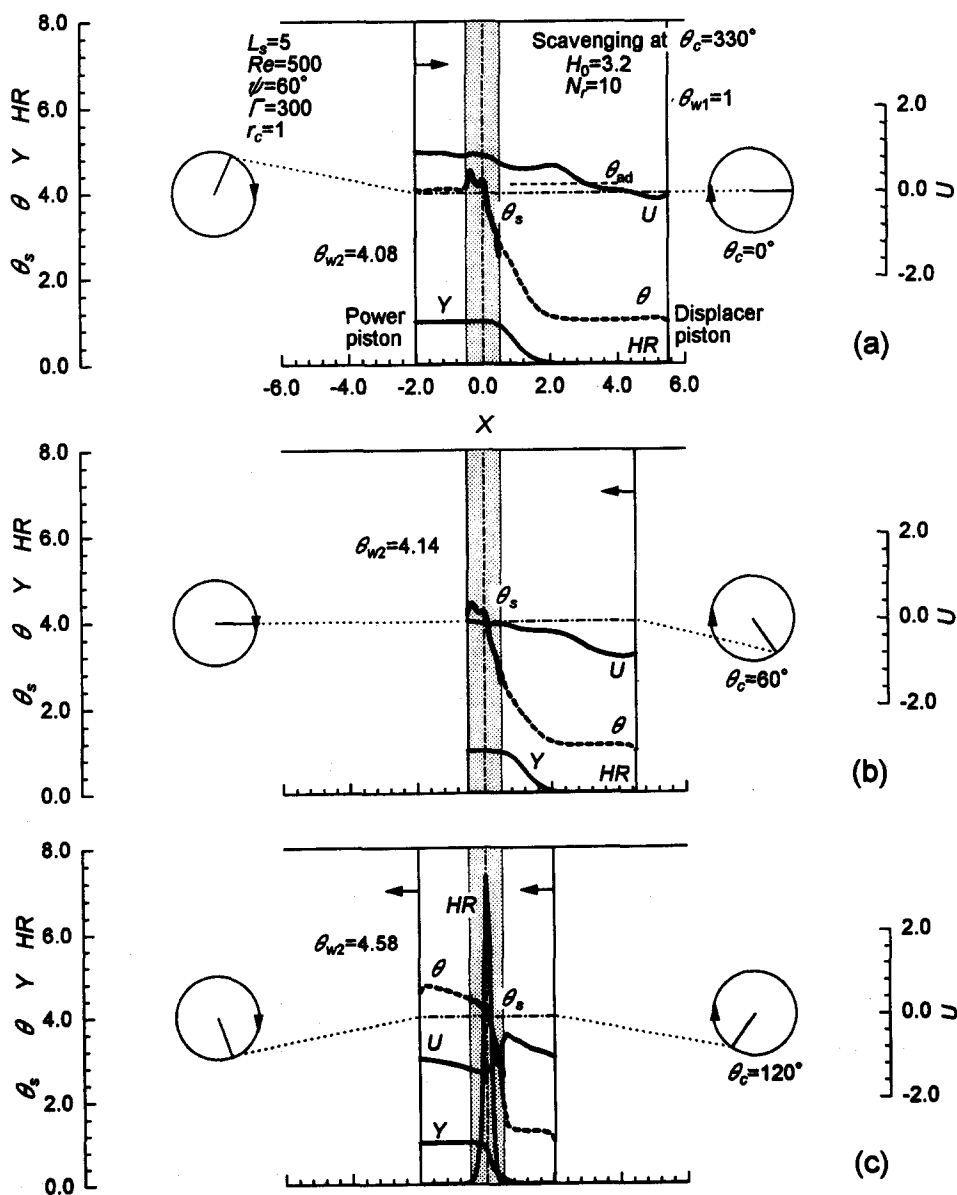


Fig. 2. (a-c)—Continued opposite.

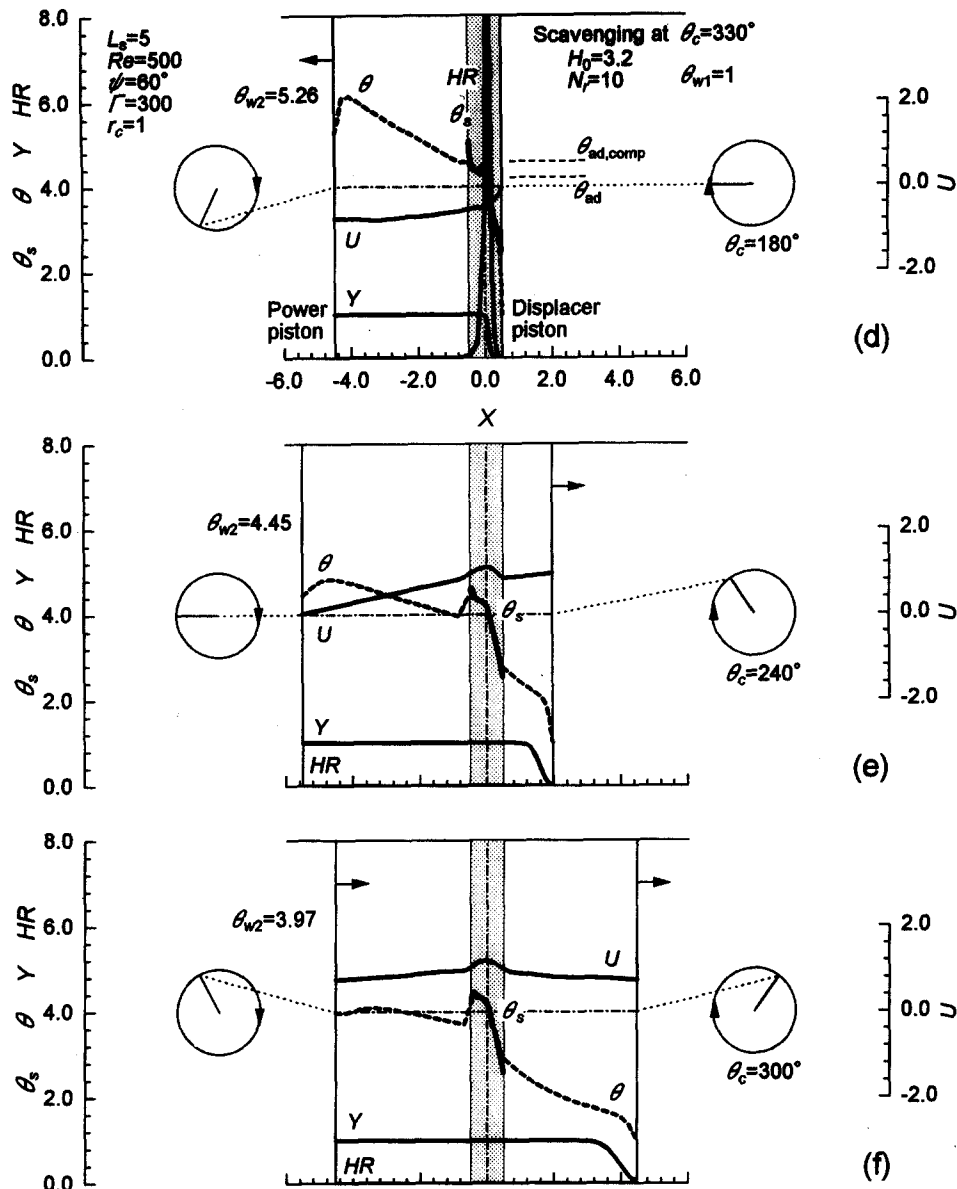


Fig. 2. Variations in profiles of gas and solid temperature, gas velocity, product mole fraction and reaction rate with respect to crank angle over one cycle of the super-adiabatic combustion engine.

and regenerated to induce the enthalpy increase of mixture in the next cycle. It is considered that the variation of the wall temperature becomes small if the heat capacity of the wall is taken into account. However, the effect to the thermal efficiency is very small since there is only a small heat loss in the power piston side.

Figure 5 shows the flammability limit for the super-adiabatic combustion engine, where the horizontal axis expresses the dimensionless heating value H_0 , while the vertical axis the conduction-radiation parameter N_r . With decreasing conduction-radiation parameter, the maximum temperature in the porous medium decreases because of the effect of the thermal diffusion by radiation. As a result, the flame is extinguished at the higher heating value with decreasing conduction-radiation parameter, as depicted by the solid line in Fig. 5. Furthermore, in the range between the solid and dashed lines, the maximum temperature in the porous medium is lower than the theoretical flame temperature in free space, though the flame is stabilized in the porous medium. Consequently, the super-adiabatic combustion is realized in the porous medium provided that the porous medium has a large absorption coefficient.

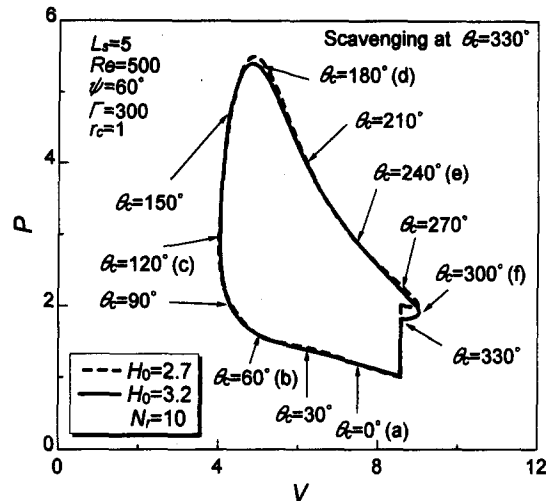


Fig. 3. P-V diagram of super-adiabatic combustion engine.

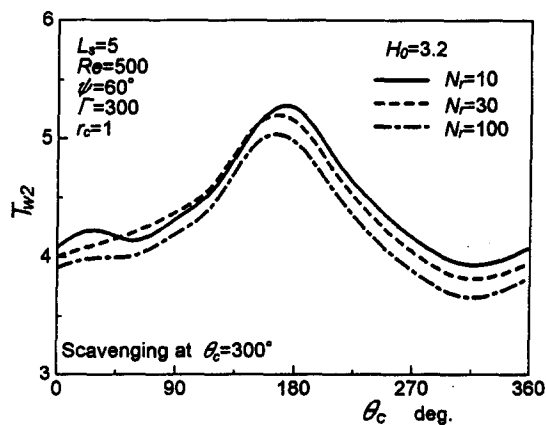


Fig. 4. Variation of wall temperature in a cycle.

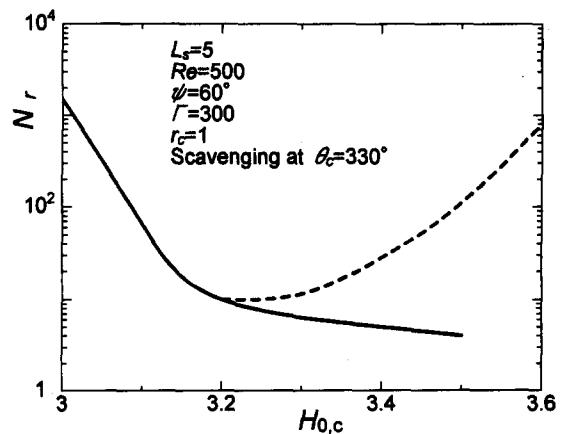


Fig. 5. Flammability limit for radiation-conduction parameter and heating value.

CONCLUSIONS

A numerical calculation has been performed on a heat engine capable of super-adiabatic combustion in a porous medium and the following results were obtained.

(1) On the basis of the super-adiabatic combustion with reciprocating flow in a porous medium, the reciprocating heat engine has a thermal efficiency of about 26% even at the flammability limit heating value.

(2) The thermal efficiency is higher than that of the conventional Otto and diesel cycles for the compression ratio of 2.

(3) Using a porous medium with a large conduction-radiation parameter, i.e a large absorption coefficient, the maximum temperature increases, resulting in the extension of the flammability limit for the super-adiabatic combustion engine.

REFERENCES

1. Commercial reports, ADTEC Co. Ltd, 1990.
2. Hanamura, K., Echigo, R. and Zhdanok, S. A., *Int. J. Heat Mass Transfer*, 1993, **36**(13), 3201-3209.
3. Hanamura, K., Miyairi, Y. and Echigo, R. *8th Int. Symp. on Transport Phenomena in Combust.* San Francisco, 1995.
4. Issiki, N. and Moriya, S., *3rd Int. Conf. on Stirling Engines*, Vol. 2, pp. 749-761, 1986.
5. Issiki, N., Moriya, S. and Fukushima, K., *4th Int. Conf. on Stirling Engines*, pp. 99-104, 1988.
6. Rallis, C., *24th Int. Energy Conv. Engng Conf.*, pp. 2395-2399, 1989.
7. Gosman, A. D., Johns, R. J. R., Tipler, W. and Watkins, A. P., *Proc. 13th CIMAC Conference*, Vol. 1-D21, pp. 27-35, Vienna, 1979.
8. Miyairi, Y., *SAE Tech. Paper Series, Int. Congress Exposition*, pp. 83-95, Detroit, 1988.